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DESIGN AND FATIGUE TESTING OF INTEGRAL ARMORED SERVO ACTUATOR MODIFIED TRUNNION

K. Wood, et al

United Aircraft Corporation

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Army Air Mobility Research and Development Laboratory

October 1974

EUSTIS DIRECTORATE POSITION STATEMENT

During previous tests of an integrated Dual Property Steel Armor CH-54B flight control hydraulic servo actuator, published as USAAMRDL-TR-73-25, several significant deficiencies were experienced during fatigue tests which caused trunnion failures and galling of the main piston and rod end threads of the servos.

This report covers the design and test of a modified trunnion assembly. The results of this effort proved to be acceptable in meeting the strength requirements of the original CH-54B trunnion.

The technical monitor for this contract was Mr. Rocco Fama of the Safety and Survivability Technical Area, Military Operations Technology Division.

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The program summarized by this report was conducted to design and establish the structural strength of a modified trunnion design for the integral armored servo actuator. The originally designed trunnion was found to have a fatigue strength below the mean strength level of comparable CH-54B parts as reported in USAAMRDL Technical Report 73-25.

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20. Continued.

The design analysis and fatigue test data show that the modified trunnion is comparable in strength to the mean strength levels of the equivalent components on the production CH-54B helicopter.

PREFACE

This report covers the design and test of a modified trunnion assembly used in the dual property steel armored servo for the CH-54B primary flight control system. The program was conducted for the Eustis Directorate under Contract DAAJ02-74-C-0013, Project 1F262205AH8801. Previous contracts DAAJ02-70-C-0051 and DAAJ02-72-C-0071 cover the design, fabrication, and testing of the integrally armored servo actuator.

Eustis Directorate technical direction was provided by Mr. R. Fama.

The principal contributors for Sikorsky Aircraft were Mr. K. Wood, Task Manager; Mr. G. Kudasch, Designer; Mr. K. Farkas, Supervisor Hydraulics Laboratory; Mr. W. Throp and Miss L. Moriarty, Cognizant Test Engineers; and Mr. S. Murich, Technician.

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BACKGROUND

Under previous contracts, DAAJ02-70-C-0051 and DAAJ02-72-C-0071, unique primary tandem servo actuators have been developed and tested for the CH-54B helicopter. The design entailed the use of dual property steel armor (DPSA) to fabricate the servo cylinder. Development and testing under these contracts have demonstrated the functional interchangeability of the DPSA servo with the production unit of the CH-54B. Structural substantiation testing completed under DAAJ02-72-C-0071 determined that the DPSA servo had a structural strength on a level with the production CH-54B servo with the exception of the trunnion assembly. Accordingly, under this contract Sikorsky redesigned and structurally tested the trunnion assembly with the goal of equaling the mean endurance strength of the equivalent production parts.

INTEGRAL ARMORED SERVO MODIFIED TRUNNION

DESIGN

The servo trunnion is a fully gimbaled device providing two degrees of motion. It reacts servo output forces, transmitting them through a stationary structure to the main transmission housing. The trunnion consists of a full ring surrounding the body of the servo. Two pins, diametrically opposed, connect the ring to the servo housing and provide one axis of rotation. Two lugs, also diametrically opposed, are displaced 90° from the pin axis and provide the second axis of rotation (see Figure 1 and Figure 2).

Failures encountered in the original trunnion included fractures both of the pin and of the lug. The pin design included a relatively thin integral flange which was bolted to the trunnion ring to maintain the pin in position. Under heavy loading, the flange tended to prevent deflection of the pin, thereby reacting part of the imposed bending moment. Fractures occurred at the change of section where the flange joined the straight section of the pin. In the modified design, the pin was fabricated as a straight diameter with a separate cover plate to retain the pin in place.

The fracture of the trunnion lug was at the change of section where the lug joined the trunnion ring. Analysis of the fracture concluded that poor machine finish and questionable shot-peening quality in the lug/ring radius were contributing factors. To strengthen the lug, the new trunnion employs a larger lug diameter with a considerably more generous radius. In addition, care was taken during fabrication to provide the proper finish and shot-peening to the radius.

In addition to the specific changes related above, a general attempt was made to utilize the full space envelope available to provide stiffening of the ring structure. Also, by more carefully contouring the I.D. of the ring to the servo housing O.D., a closer tolerance fit was maintained, resulting in a minimum unsupported area of the pin as it traverses between the ring and servo housing.

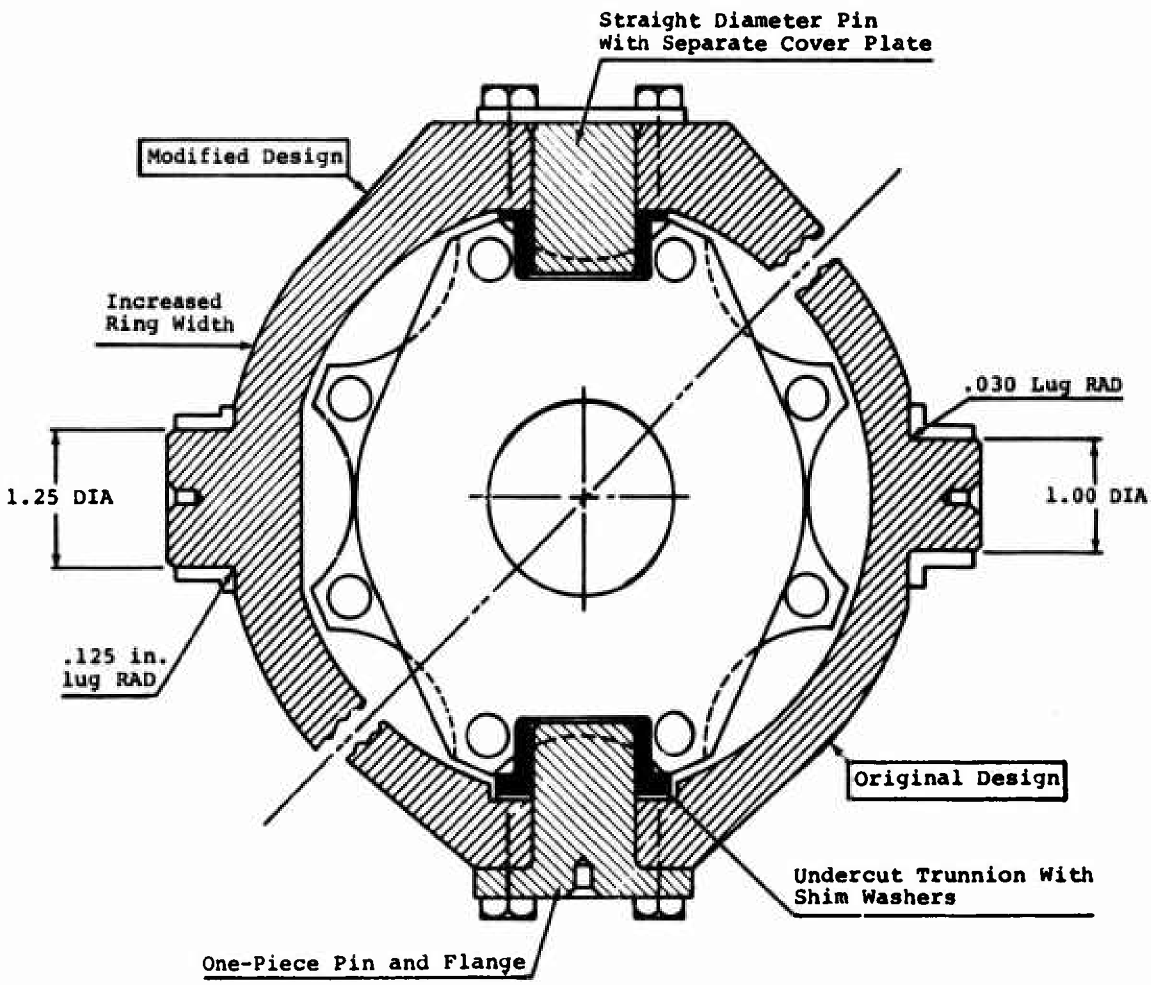

Figure 1. DPSA Servo Trunnion - Modified Versus Original Design.

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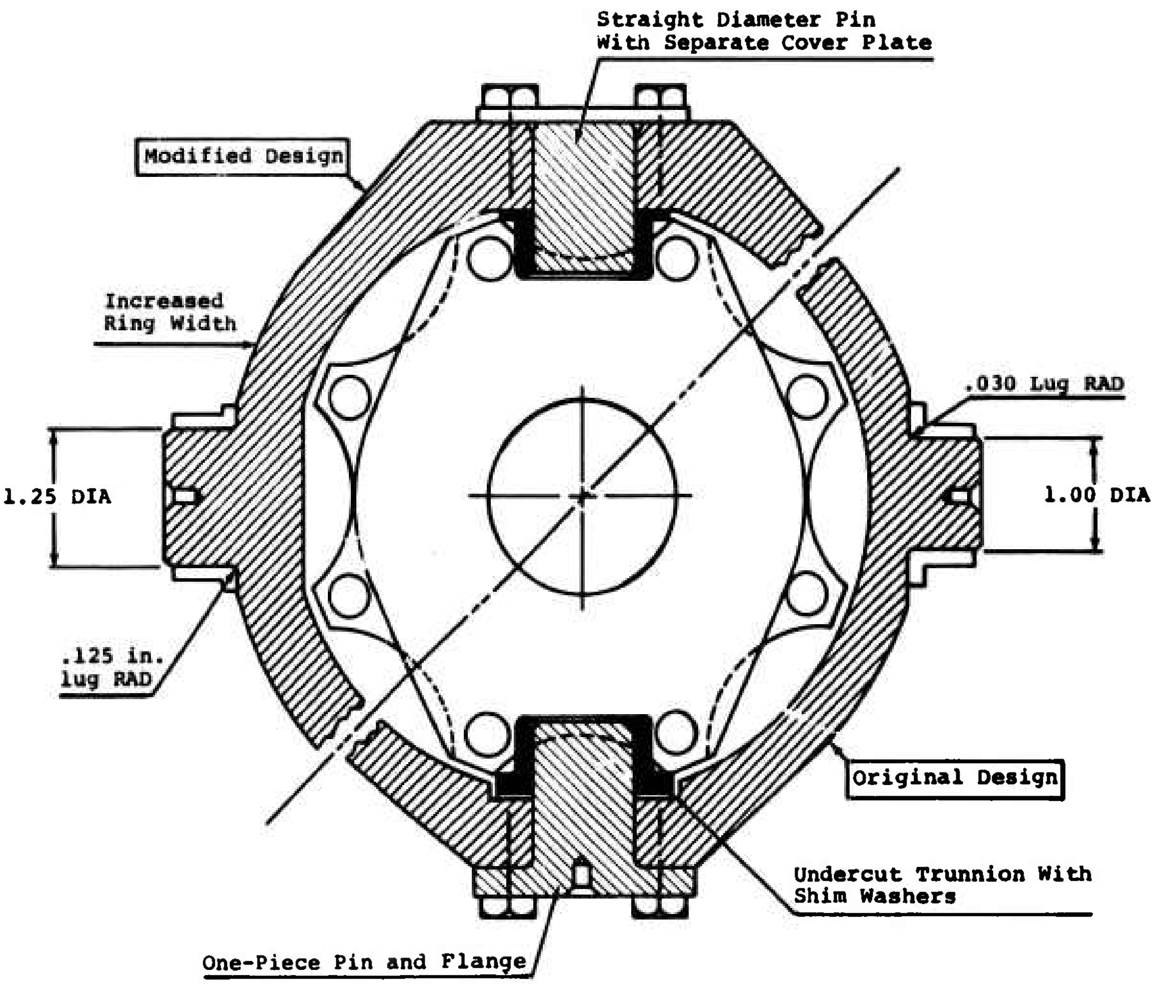

Figure 1. DPSA Servo Trunnion - Modified Versus Original Design.

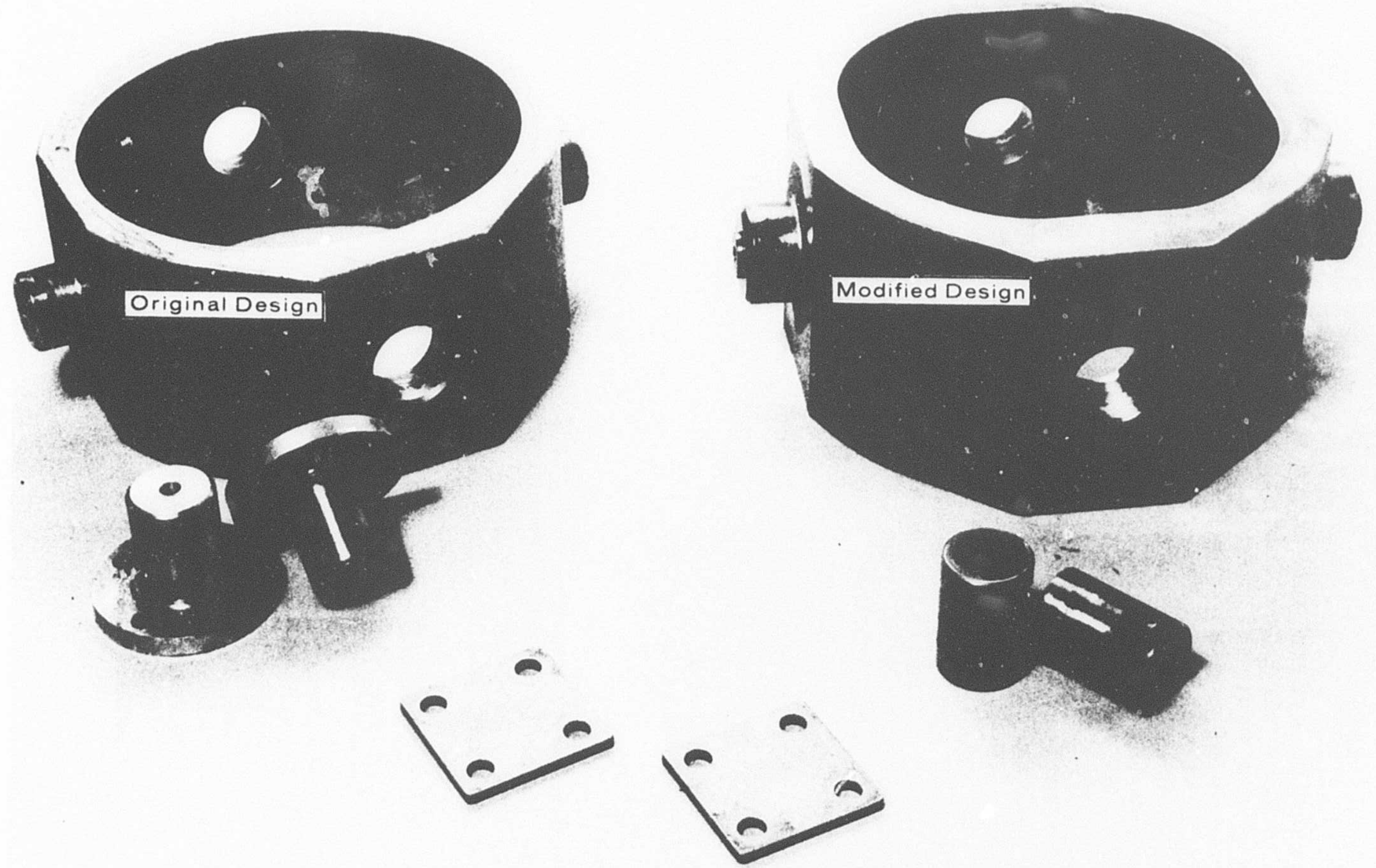

Figure 2. Side-by-Side Comparison - Modified Versus Original Trunnion.

STRESS ANALYSIS

An iterative static and fatigue stress analysis was made on the trunnion ring lug during the design phase. In this fashion the most influential dimensions were utilized within the bounds of the physical space available and without adding inefficient weight. The static and fatigue strength were determined on both an absolute and a relative basis, the latter comparing the original trunnion design with the modified design.

The theoretical fatigue analysis for the two designs considered published small-specimen fatigue strength data for the materials used, corrected by size effect and stress concentration factors. An S/N curve for the mean endurance limit of each design was constructed upon which the actual fracture points of the original lug design were superimposed. The theoretical curve for the old design was then corrected to cause it to pass through the test points, and that for the new design was corrected in a like manner.

The new design reflects an improvement in theoretical static strength of 1.85. The new design also increases the endurance limit by a factor of 1.89, causing the 3σ limit to exceed the design limit load for the CH-54B. The detailed analysis is presented in the appendix.

TEST PROGRAM

One modified trunnion was fabricated and assembled with one of the three previously fabricated CH-54B DPSA servo actuators. Testing was accomplished in accordance with the following test plan.

Test Plan

Test Conditions

(1) Ambient Temperature: $70° \pm 20°F$
(2) Fluid Temperature: $80° \pm 30°F$
(3) System Filtration: 5 microns nominal, 15 absolute (per MIL-F-8815)
(4) Fluid: MIL-H-5606A
(5) Operating Pressure: First Stage 0-4500 psi
Second Stage 0-4500 psi
(6) Normal Operating Return Pressure: 30 psi

Deflection Test

The servo assembly with the modified trunnion was installed in the fatigue test fixture. Pressures from 0 to 4500 psi were applied in 500-psi increments, with deflection data recorded at each point. Data for two complete pressure cycles from 0 to 4500 psi extend to 4500 psi retract to 0 were recorded.

Fatigue Test

The servo assembly with the modified trunnion was subjected to a fatigue test of 10^6 pressure cycles. The fatigue fixture constrained the servo at approximately mid-stroke. Pressure cycles of 0 psi to 4500 psi and back to 0 psi were applied to both stages at the rate of about four per second. The servo was tested with the servo valve in the hard-over extend position.

Trunnion Deflection Versus Load Test Results

The trunnion deflection versus load test setup is shown in Figure 3, with detailed photographs, Figure 4 and Figure 5, showing a close-up of the trunnion installation and the deflection measurement system. The trunnion spring rate was found to be 1,692,000 lb/in. in the retracting direction and 1,450,000 lb/in. in the extending direction. A complete listing of trunnion deflection versus load is contained in Table I, and a plot is given in Figure 6.

Trunnion Fatigue Test Results

The trunnion fatigue test setup is shown in Figure 7, with detailed photographs, Figure 8 and Figure 9, showing the load-monitoring instrumentation and load-cycling apparatus. Fatigue data analysis is presented here for the servo trunnion assembly, of which one test item was tested without failure. Data is compared with fatigue test data for the main rotor servos used on the CH-54B helicopter and with fatigue test data for the original trunnion design.

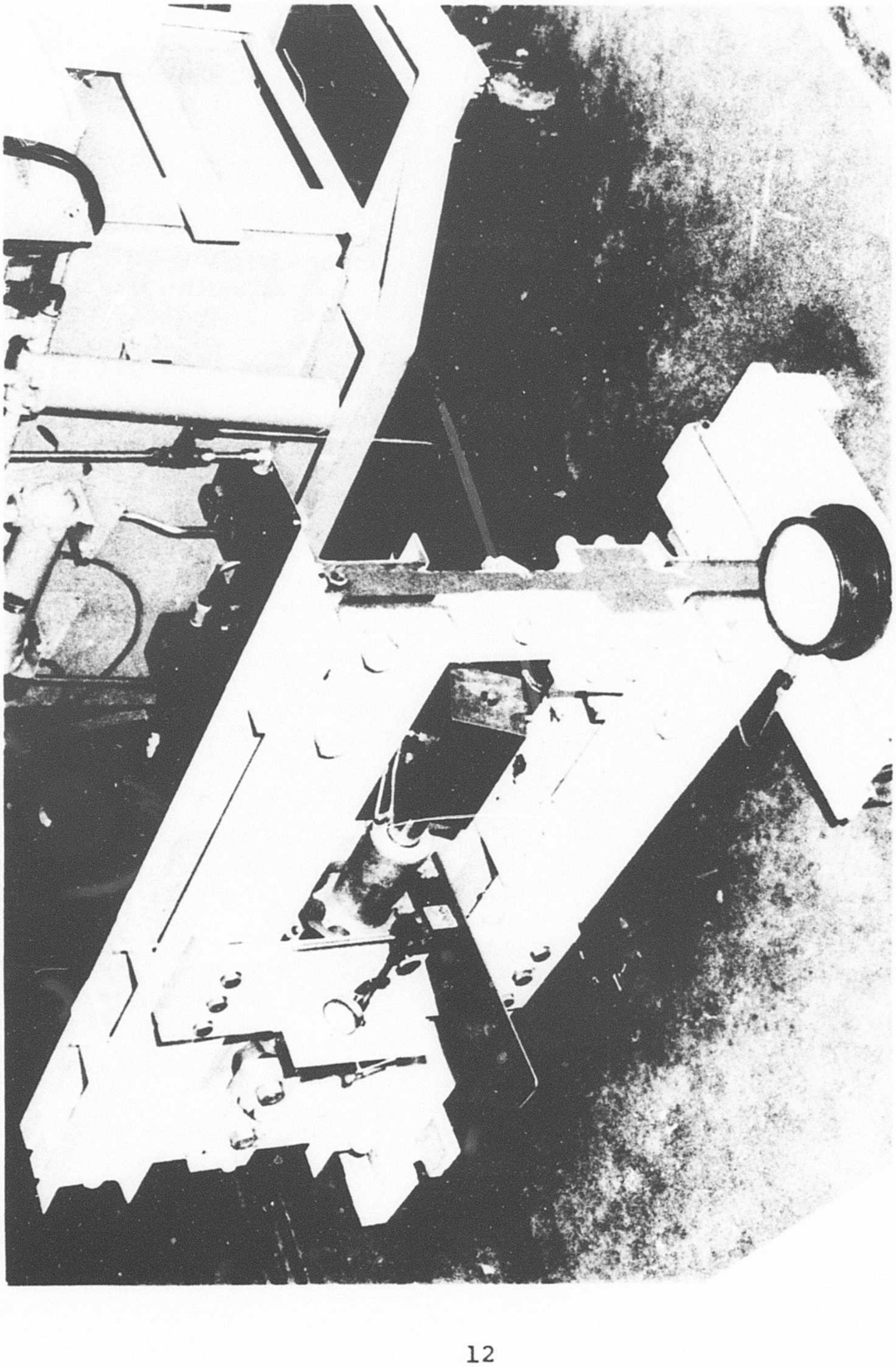

Figure 3. Modified Trunnion Deflection Test Setup.

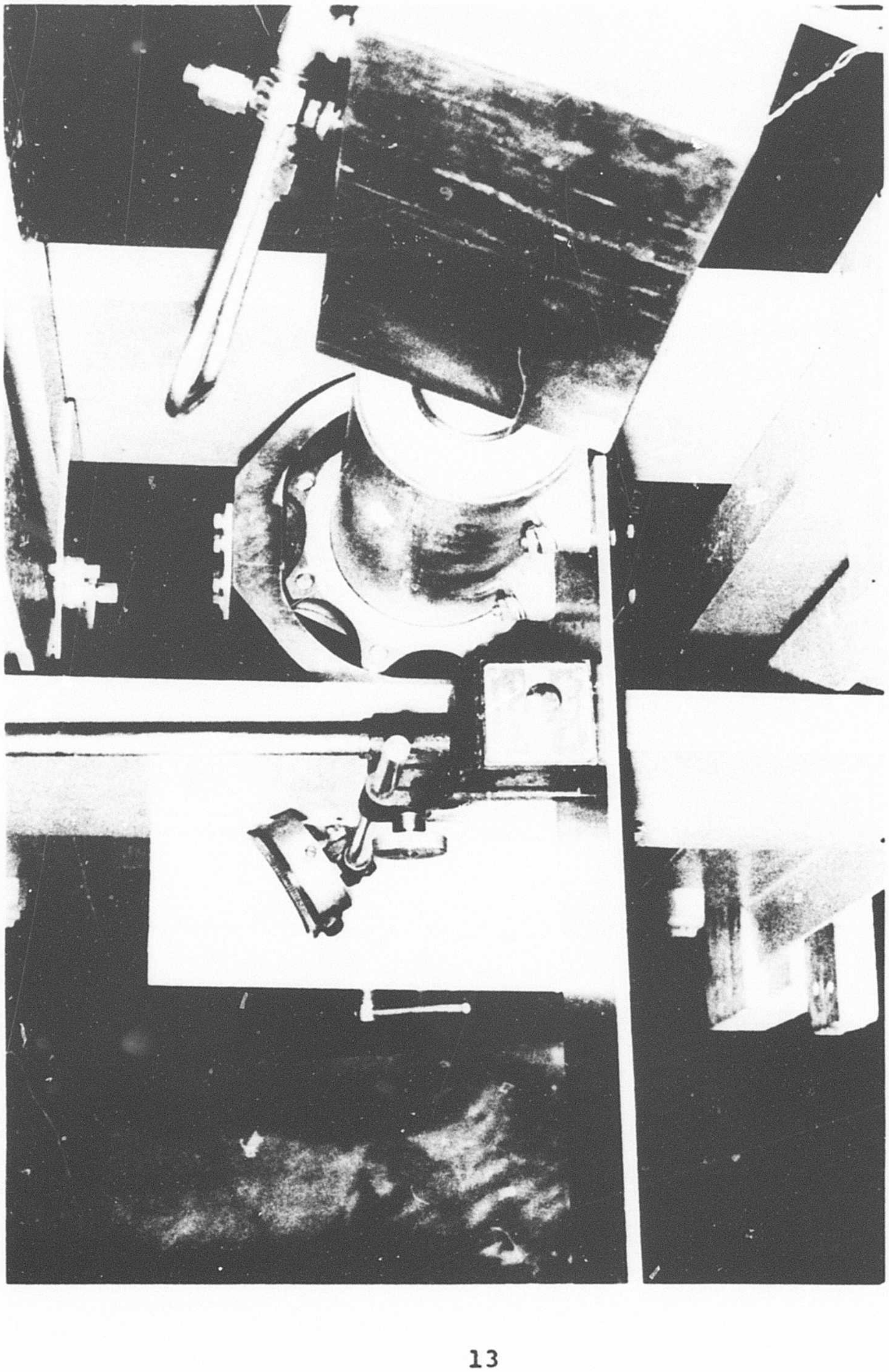

Figure 4. Close-Up of Deflection Measurement Apparatus.

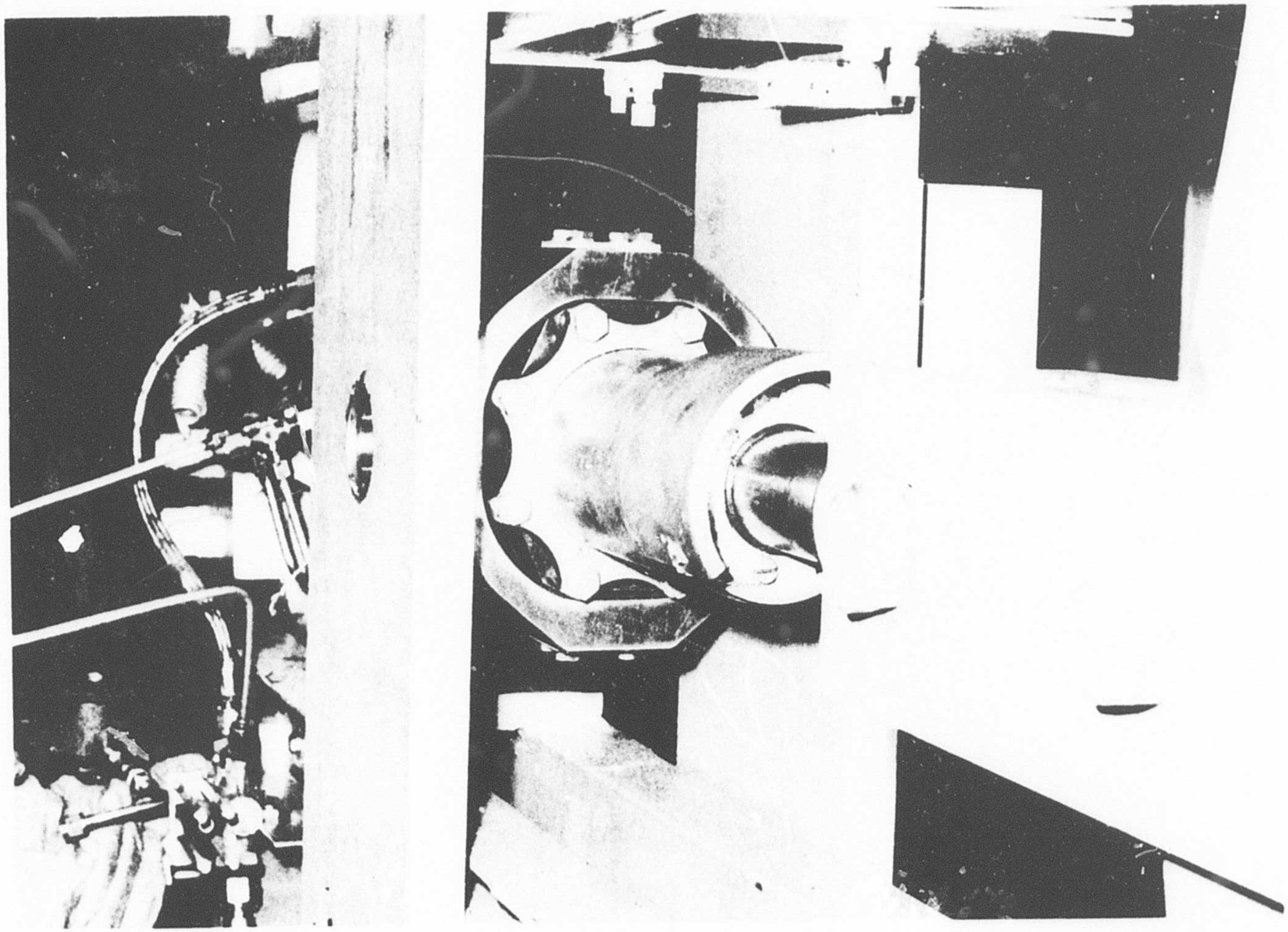

Figure 5. Close-Up of Modified Trunnion Installation.

TABLE I. Modified Trunnion Deflection Characteristics				
Applied Pressure (psig)	Extension (in.x10^3)	Retraction (in.x10^3)	Extension (in.x10^3)	Retraction (in.x10^3)
0	0.0	1.5	1.5	1.5
500	2.0	-2.0	7.0	-2.0
1000	4.5	-4.5	9.5	-4.5
1500	7.0	-6.5	12.0	-7.0
2000	9.5	-9.0	14.5	-9.0
2500	12.0	-11.0	16.5	-11.0
3000	14.5	-13.0	18.0	-13.5
3500	17.0	-15.0	19.0	-15.0
4000	19.0	-16.5	20.5	-17.0
4500	21.5	-18.0	22.5	-18.5
4000	20.0	-16.0	21.0	-16.5
3500	18.0	-14.0	19.5	-14.5
3000	16.5	-12.0	17.5	-12.5
2500	14.5	-11.0	15.0	-11.5
2000	12.5	-9.5	13.0	-10.0
1500	10.0	-8.0	10.5	-8.0
1000	7.5	-6.5	7.5	-6.5
500	4.5	-4.5	4.5	-4.5
0	1.0	1.0	0.5	0.5

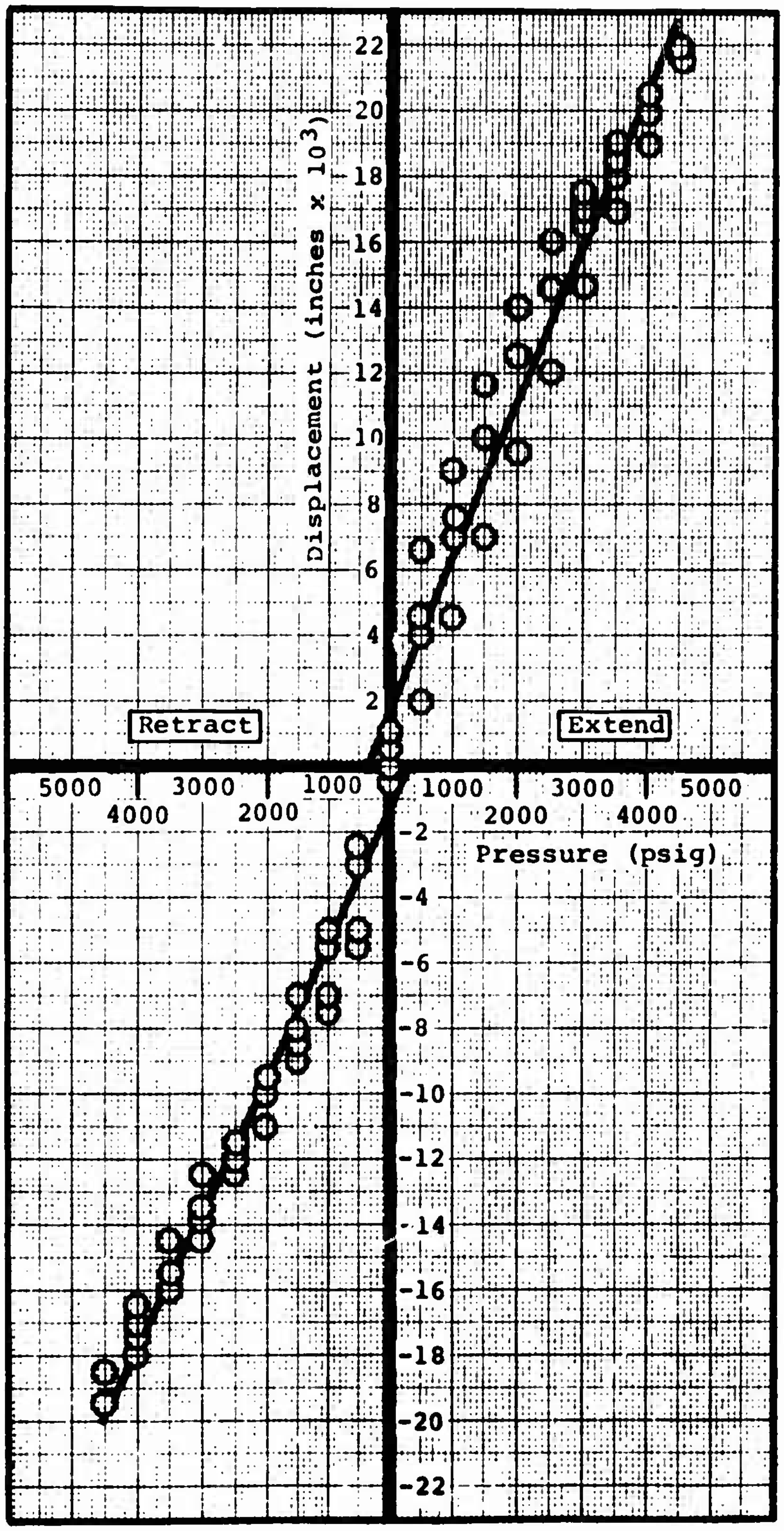

Figure 6. Modified Trunnion Deflection Versus Servo Pressure.

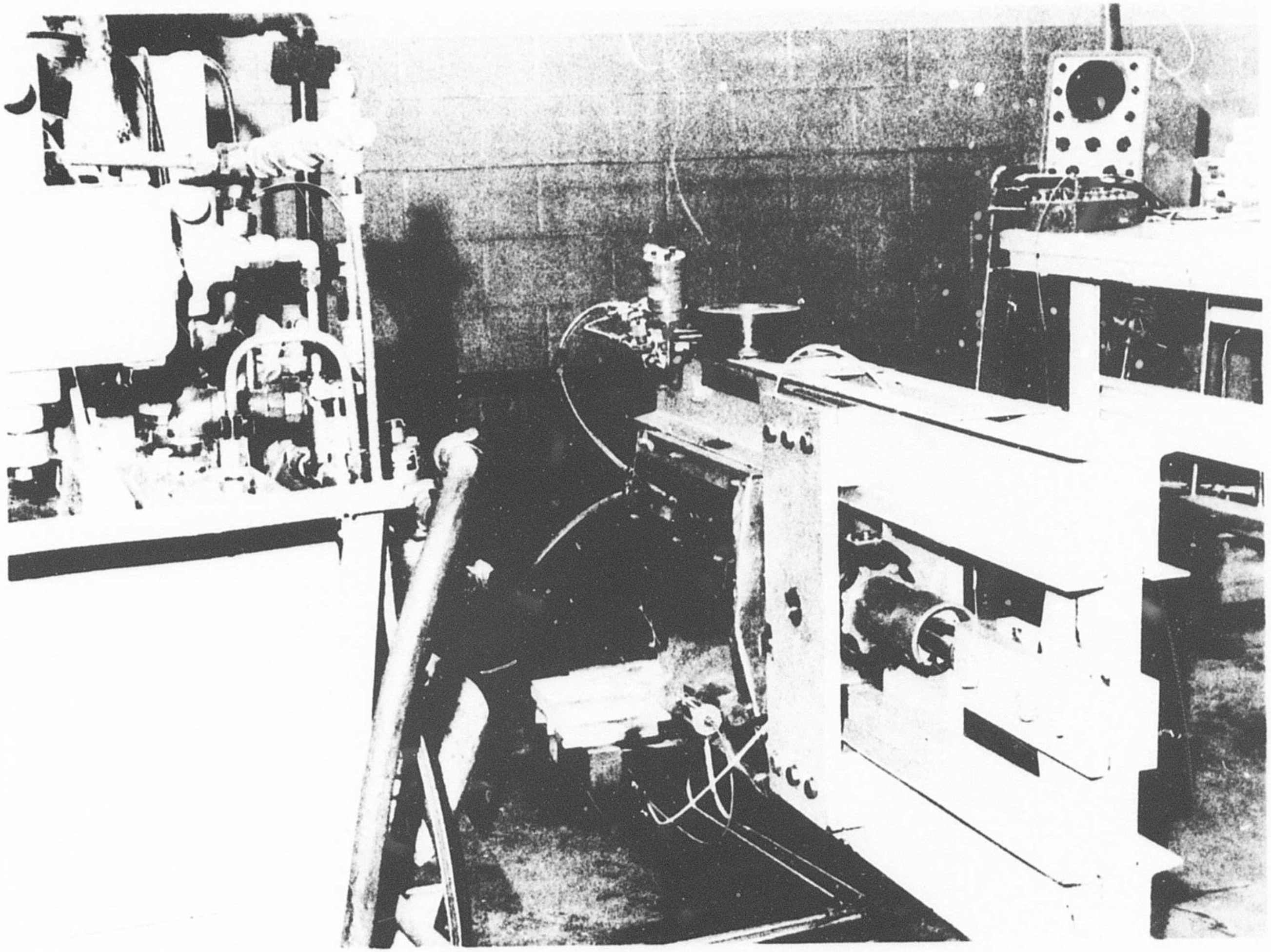

Figure 7. Modified Trunnion Fatigue Test Setup.

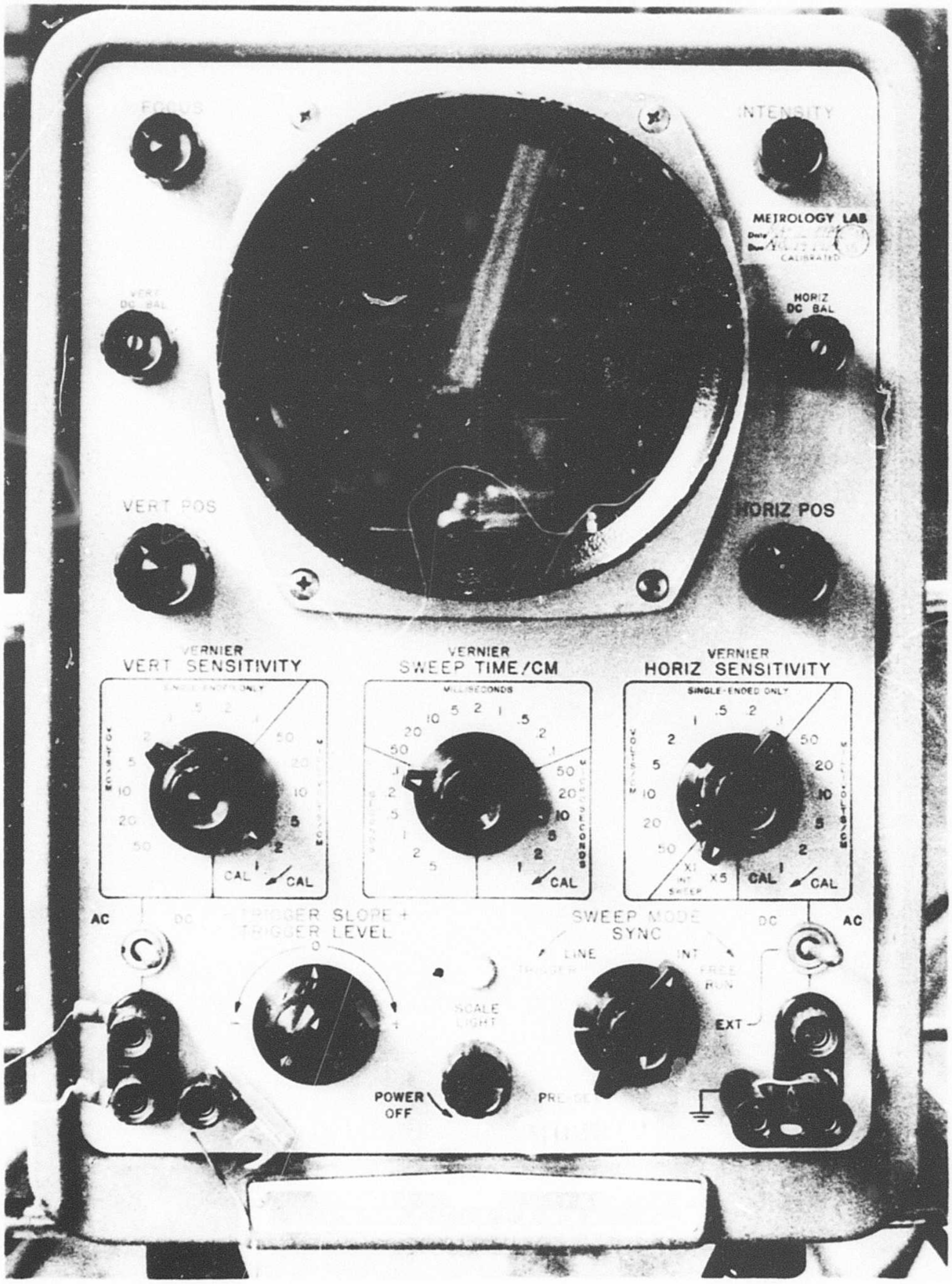

Figure 8. Modified Trunnion Fatigue Test Load-Monitoring Apparatus.

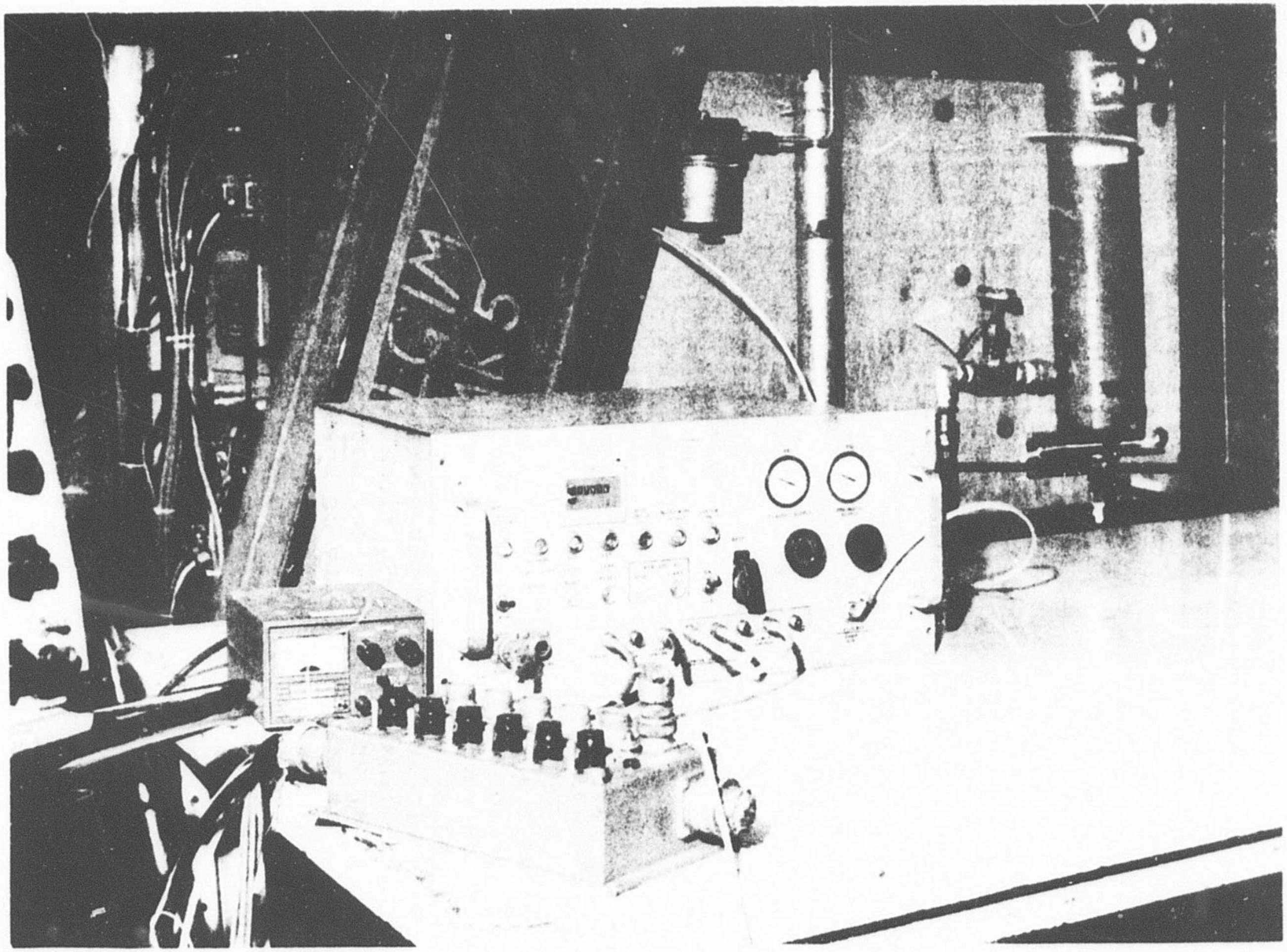

Figure 9. Modified Trunnion Fatigue Test Fluidic Cycling Apparatus.

Data Analysis

The modified trunnion sustained a runout and completed 1,000,000 pressure cycles of 0 to 4500 to 0 psi at a rate of approximately four cycles per second. As the servo has a total piston area of 7 square inches, the maximum axial force on the servo was 31,500 lb, which results in an equivalent 15,750 lb vibratory about a mean axial force of 15,750 lb. Data points to construct the S/N curve for the trunnion were calculated as follows:

It is assumed that for a given material and chafing condition, a single curve shape is applicable regardless of the strength or configuration of the test article. The curve shape for metals is defined as

$$E = F_{vib} \left(1 + \frac{\beta}{N^{\zeta}}\right)^{-1} \qquad (1)$$

where E = endurance limit of material
F_{vib} = applied vibratory load
β = curve shape parameter
ζ = curve shape parameter
N = component life (cycles x 10^{-6}) to crack detection at the applied load.

The curve shape parameter values for β and ζ for use in equation (1) have been established by fitting the equation to large quantities of available test data acquired at a sufficient number of test levels to provide information on curve shapes. Such data is obtained from MIL-HDBK-5, Alcoa Test Results, Sikorsky Test Data, and other similar sources. It should be noted that E and F_{vib} may be specified in either units of stress (psi) or loading (lb, lb-ft, psi, etc.) as long as the units in any one application are consistent and a linear relationship between stress and load has been assumed.

The shape of the S/N curve for steel without chafing is

$$E_T = F_{vib} \left(1 + \frac{\beta}{N^{\zeta}}\right)^{-1} \qquad (2)$$

where E_T = endurance limit, based on test data for the modified trunnion

F_{vib} = applied vibratory load (lb)

β = 0.0323

ζ = 1.0

N = number of cycles x 10^{-6} on the modified trunnion at which the sample fractured, or at which the test was stopped (runout).

The runout point obtained will be considered a fracture for the purposes of constructing an S/N curve and working curve shape. The above equation yields

$$E_T = 15,257 \text{ lb} \tag{3}$$

The variability of steel in fatigue has been shown to be 10%. Applying a confidence level factor of three times the variability, the working curve value for the fatigue endurance limit is then

$$\begin{aligned} E_{TW} &= .70 \times 15,257 \\ &= 10,680 \text{ lb} \end{aligned} \tag{4}$$

The S/N curves for the modified armored servo trunnion, original trunnion, and CH-54B production mounting assembly are shown in Figure 10. The fatigue strength of the modified trunnion exceeded that of the production mounting assembly based on a single test data point.

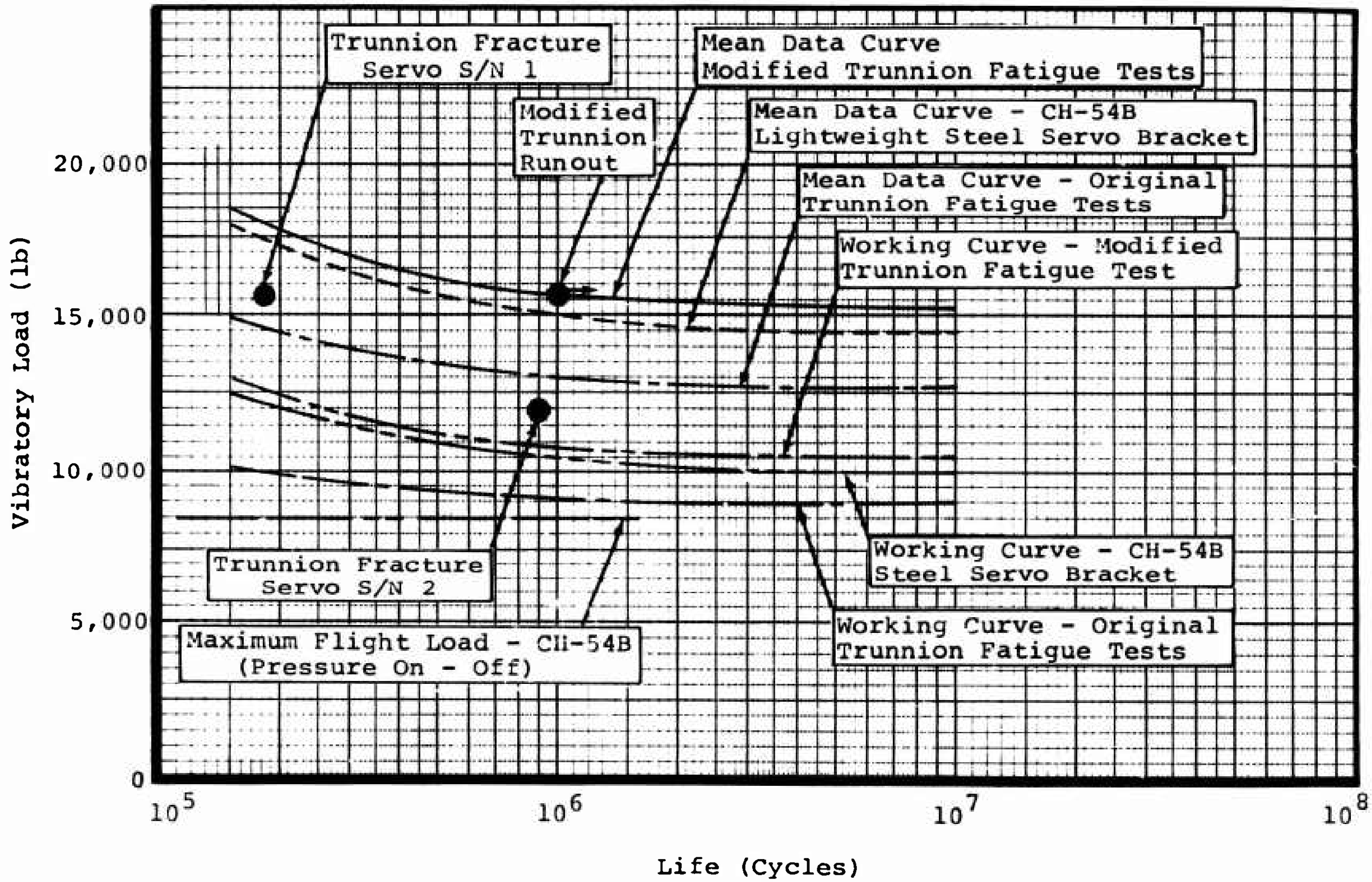

Figure 10. Modified Trunnion Mean and Working S/N Curves.

CONCLUSIONS

1. The modified trunnion satisfactorily met the structural strength requirements defined in the test plan.

2. The stiffness of the trunnion assembly was determined. The value is such that stable operation of the DPSA servo assembly should be realized.

3. The modified trunnion test reported herein and all parts of the DPSA servo assembly previously tested were shown to have endurance limits equivalent to the comparable components of the production CH-54B primary servo.

4. Based on only a single data point, the modified trunnion is considered airworthy for limited flight usage on the CH-54B helicopter. Test of additional specimens and/or higher load levels is required to lift this limitation.

APPENDIX
STRESS ANALYSIS - MODIFIED TRUNNION

SUMMARY

Fatigue testing of the trunnion reported under Contract DAAJ02-72-C-0071 showed that the critical portion of the structure was the support lug. (See Figure 11.) The structural strength of the lug was shown to be lower than that of the equivalent CH-54B production part. The critical load of the spectrum is imposed by the "OFF-ON-OFF" cycles of the 3000-psi hydraulic system.

A comparative stress analysis was conducted to determine what changes were required in order to meet a no-damage criterion (infinite life) under the critical load spectrum. Small specimen fatigue data from SER 50586 were used to predict S/N characteristics of the original and modified designs. Empirical corrections were then made by using full-scale test results to adjust the mean fatigue curves.

The analysis indicated that the increases in lug diameter and reduction of stress concentrations should result in the structure meeting an infinite life criterion.

ANALYSIS

The geometries of the original and the modified lug designs are given in Figure 11.

The static stress analysis is presented in Table II. The fatigue analysis was based on SER 50586.[1] The theoretical stress concentration factor, K_t, was obtained from Figure 65 of STRESS CONCENTRATION DESIGN FACTORS[2] by using the calculations in Table III.

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[1] Degnan, W. G., FATIGUE PROPERTIES AND ANALYSIS, Sikorsky Aircraft SER 50586, revised February, 1971.

[2] Peterson, R. E., STRESS CONCENTRATION DESIGN FACTORS, John Wiley & Sons, 1953.

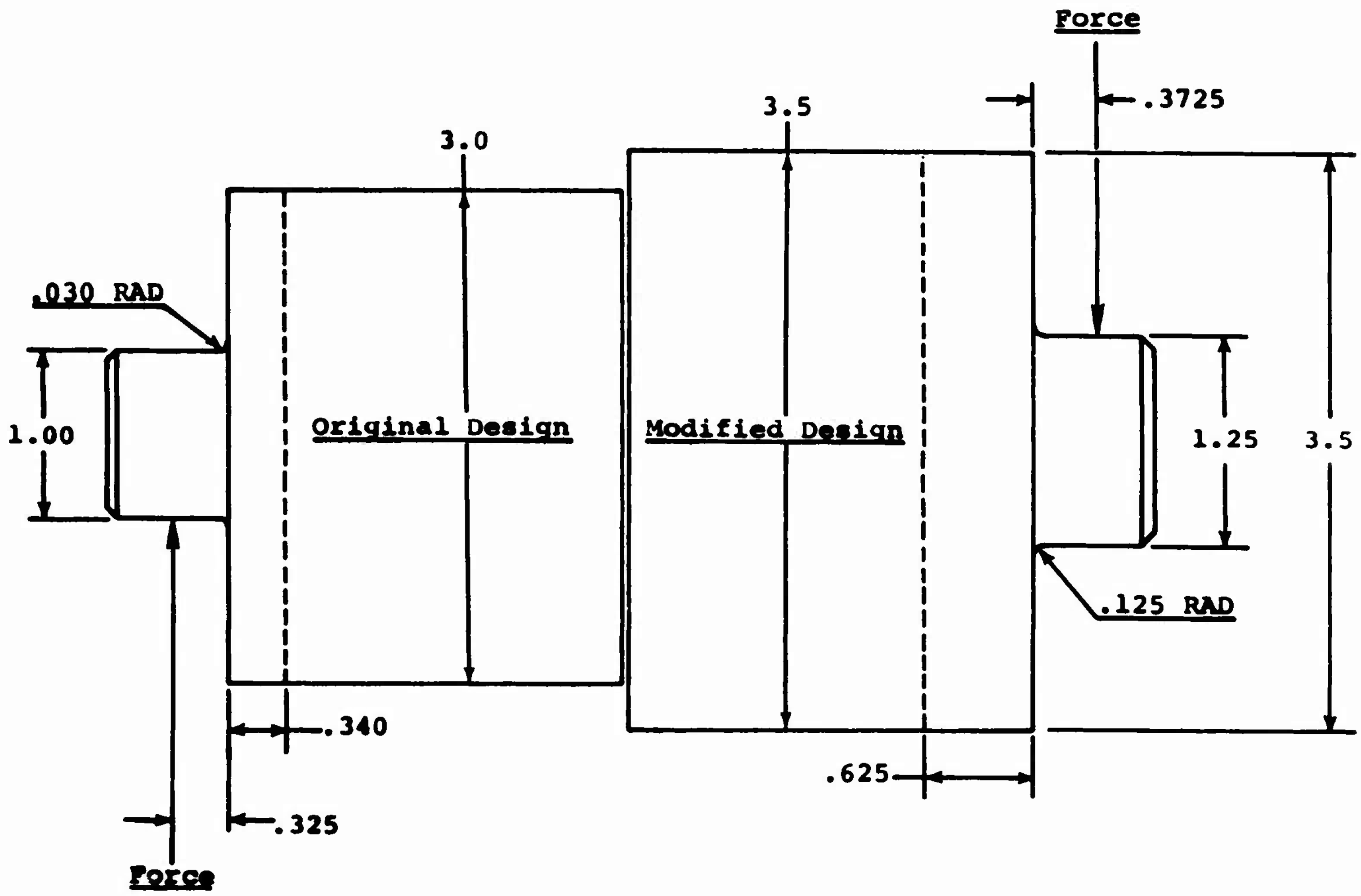

Figure 11. Lug Design Geometry - Original Versus Modified.

TABLE II. Static Stress Analysis		
Parameter	**Original Trunnion**	**Modified Trunnion**
Effective area of piston, A	7.08 in.^{2}	7.08 in.^{2}
Design limit pressure load, p	3000 psig	3000 psig
Force on servo, F = pA	21,240 lb	21,240 lb
Force on each lug, f = F/2	10,620 lb	10,620 lb
Moment arm, L	.325 in.	.3725 in.
Radius of lug, r	.4998 in.	.6249 in.
Area moment of inertia, $I=\frac{\pi r^4}{4}$	.048998 in.^{4}	.11975 in.^{4}
Moment, M=fL	3,452 in.-lb	3,956 in.-lb
Nominal max stress, in bending, $\sigma_b = \frac{Mr}{I}$	35,207 psi	20,643 psi
Area of lug, $A_{lug} = \pi r^2$	.7848 in.^{2}	1.2268 in.^{2}
Shear stress, $\sigma_s = F/A_{lug}$	13,532 psi	8,657 psi
Ultimate shear stress, F_{su}	95,000 psi	95,000 psi
Ultimate bending stress, F_{bu}	300,000 psi	300,000 psi
Ultimate margin of safety $MS_u = \dfrac{1}{1.5 \quad \frac{b^2}{F_{bu}} + \frac{s}{F_{su}}^2}$	2.61	4.84

TABLE III. Calculation of the Theoretical Stress Concentration Factor		
Parameter	**Original Trunnion**	**Modified Trunnion**
Fillet radius, r'	.030 in.	.125 in.
Diameter of lug, d	.9995 in.	1.24975 in.
Approximate diameter of base, D	2.0 in.	1.85 in.
r/d	.030015	.10002
D/d	2.001	1.4803
K_t	2.56	1.67

Figure 12 was obtained from figures (3.2.01 and 3.1.02) of SER 50586[1] using the calculations shown in Table IV and Table V. The lower four curves of Figure 12 were plotted in Figure 13 with the ordinate changed from stress to the corresponding vibratory force in the servo. Values of the curve points are given in Table VI. In addition, the two test points for the old design were plotted in Figure 13. In Figure 14, the curves from Figure 13 have been adjusted by the ratio necessary to cause the mean curve of the old design to pass through the test points. Values of curve points are given in Table VII.

The design limit vibratory load is also plotted on Figure 14. It can be seen from Figure 14 that the original design would have a finite life under limit load, whereas the modified design is unlimited.

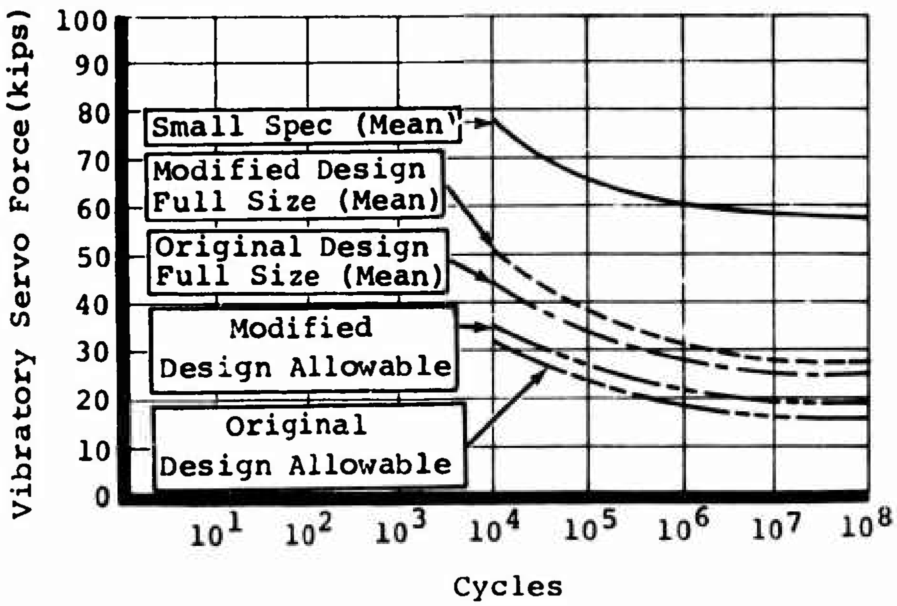

Figure 12. Theoretical Vibratory Stress Versus Life, Integrally Armored Servo Trunnion.

Table IV. Calculations of Original Design S/N Curve

Parameter	10^4 Cycles	10^5 Cycles	10^6 Cycles	10^7 Cycles	10^8 Cycles
R	0.0	0.0	0.0	0.0	0.0
K_t	2.56	2.56	2.56	2.56	2.56
K_f	1.6	1.75	1.9	2.1	2.1
K_s	1.0	1.0	1.0	1.0	1.0
F_s	.9	.9	.9	.9	.9
F_r	.7	.7	.7	.7	.7
Small Spec (mean)	78,000 psi	67,000 psi	59,000 psi	59,000 psi	59,000 psi
Full Size (mean)	43,875 psi	34,457 psi	27,947 psi	25,286 psi	25,286 psi
Design (allowable)	30,713 psi	24,120 psi	19,563 psi	17,700 psi	17,700 psi

Table V. Calculations of Modified Design S/N Curve

Parameter	10^4 Cycles	10^5 Cycles	10^6 Cycles	10^7 Cycles	10^8 Cycles
R	0.0	0.0	0.0	0.0	0.0
K_t	1.67	1.67	1.67	1.67	1.67
K_f	1.4	1.55	1.7	1.9	1.9
K_s	1.0	1.0	1.0	1.0	1.0
F_s	.9	.9	.9	.9	.9
F_r	.7	.7	.7	.7	.7
Full Size (mean)	50,143 psi	38,903 psi	31,235 psi	27,947 psi	27,947 psi
Design (allowable)	35,100 psi	27,232 psi	21,865 psi	19,563 psi	19,563 psi

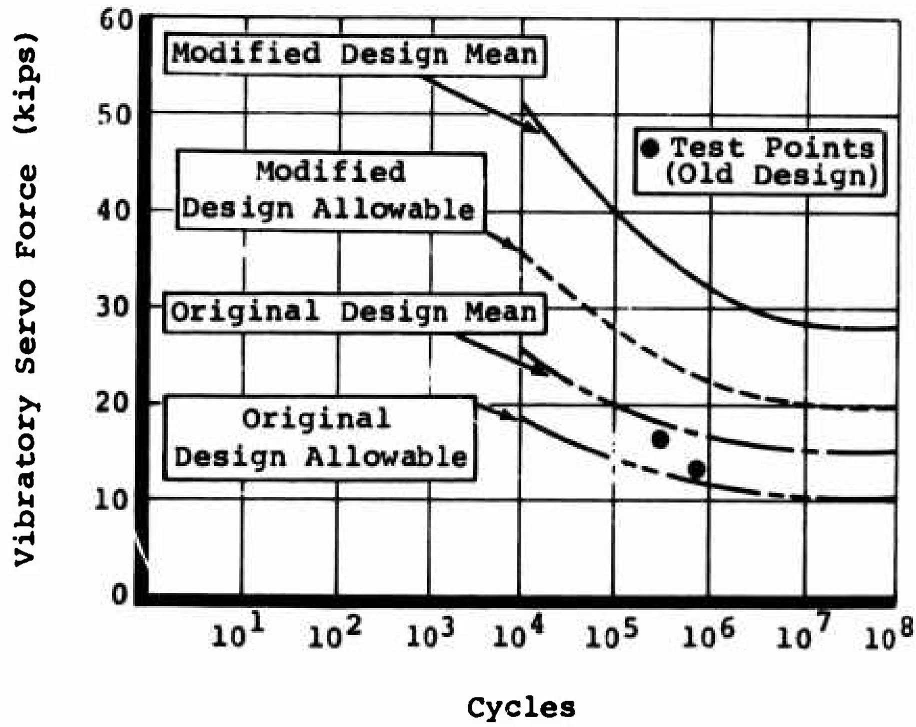

Figure 13. Theoretical Vibratory Load Versus Life, Integrally Armored Servo Trunnion.

$$F_{vib} = \left(\frac{2}{L} \quad \frac{I}{r} \right) \sigma_{VIB}$$

$$\text{Original} \quad F_{vib} = .6033 \quad \sigma_{VIB}$$

$$\text{Modified} \quad F_{vib} = 1.0289 \quad \sigma_{VIB}$$

Table VI. Conversion of S/N Curve to Load Ordinate

Parameter	Load @ 10^4 Cycles (lb)	Load @ 10^5 Cycles (lb)	Load @ 10^6 Cycles (lb)	Load @ 10^7 Cycles (lb)	Load @ 10^8 Cycles (lb)
Original Design (Mean)	24,470	20,788	16,860	15,255	15,255
Original Design (Allowable)	18,530	14,552	11,802	10,678	10,678
Modified Design (Mean)	51,592	40,027	32,138	28,755	28,755
Modified Design (Allowable)	36,114	28,019	22,497	20,128	20,128

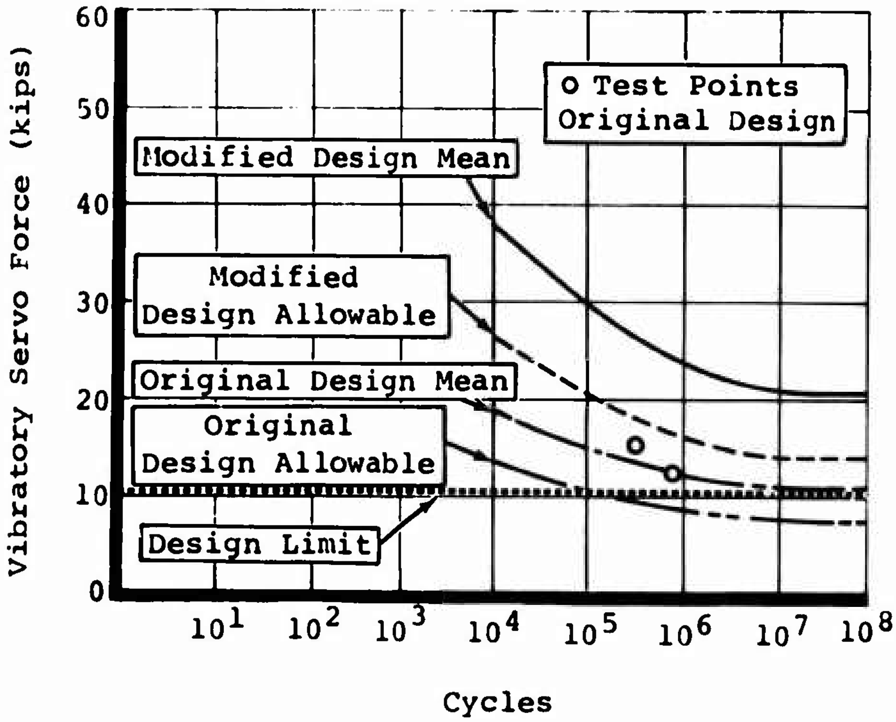

Figure 14. Vibratory Load Versus Life, Integrally Armored Servo Trunnion Corrected for Empirical Data.

Table VII. Adjustment of S/N Curve To Fit Fracture Test Data

Parameter	Load @ 10^4 Cycles (lb)	Load @ 10^5 Cycles (lb)	Load @ 10^6 Cycles (lb)	Load @ 10^7 Cycles (lb)	Load @ 10^8 Cycles (lb)
Original Design (Mean)	17,983	15,277	12,390	11,211	11,211
Original Design (Allowable)	13,618	10,694	8,673	7,847	7,847
Modified Design (Mean)	37,915	29,416	23,618	21,132	21,132
Modified Design (Allowable)	26,540	20,591	16,533	14,792	14,792

LIST OF SYMBOLS

A	effective area of piston, sq in.
A_{lug}	area of lug, sq in.
D	approximate diameter of base, in.
d	diameter of lug, in.
E	endurance limit of material, cycles
E_T	endurance limit based on test data for the modified trunnion, cycles
E_{TW}	working endurance limit based on test data for the modified trunnion, cycles
F	force on servo, lb
F_{bu}	ultimate bending stress, psi
F_r	stress reduction factor for reliability
F_s	stress reduction factor for size effect
F_{su}	ultimate shear stress, psi
F_{vib}	applied vibratory load, lb
f	force on lug, lb
I	area moment of inertia, $in.^4$
K_f	stress reduction factor for fatigue
K_s	surface finish factor
K_t	theoretical stress concentration (geometric)
L	moment arm, in.
M	moment, in.-lb
MS_u	ultimate margin of safety
N	number of cycles x 10^{-6}
p	design limit pressure load, psig

LIST OF SYMBOLS (continued)

R	ratio of minimum stress/maximum stress
r	radius of lug, in.
β	curve shape parameter
ζ	curve shape parameter
σ_b	nominal maximum stress in bending, psi
σ_s	shear stress, psi
σ_{VIB}	applied vibratory load,psi